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RESEARCH MEMORANDUM

VIBRATION SURVEY OF NACA 24-INCH SUPERSONIC AXIAL-FLOW COMPRESSOR

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SUMMARY

Vibration investigations were made of two blade cascades in wind tunnels and of blades operated in the N&CA 24-inch supersonic compressor. The results showed that the blade vibrations were present at all tunnel and compressor air velocities and were influenced primarily by tunnel design, simulated centrifugal loading, surging, angle of attack, and possible critical Mach numbers.

The highest stress amplitude observed in the tunnel tests was ±39,600 pounds per square inch, single amplitude, at a Mach number of 0.81. The maximum recorded stress in the operating compressor was ±23,250 pounds per square inch, single amplitude, at a Mach number of 0.714; in the supersonic range, the maximum recorded stress was ±14,500 pounds per square inch, single amplitude, at a Mach number of 1.41.

INTRODUCTION

An axial-flow compressor that utilizes supersonic shockwaves to convert velocity into pressure is very promising aerodynamically because it produces a high pressure rise per stage while maintaining high mass flow and good over-all efficiency (reference 1). Mechanically, however, the supersonic compressor produces many problems because of the high peripheral speeds and the extreme susceptibility to vibration resulting from its relatively thin blades.

For the NACA 24-inch supersonic compressor, the originally proposed blades were 3 inches long, only 0.048 inch thick, and had approximately $2\frac{1}{2}$-inch chords. The use of a shroud to restrain the vibration amplitude of the thin blades was planned, but the attachment of the shroud presented a very difficult problem. Preliminary investigations of shrouds, both welded and riveted, were made at the

NACA Langley laboratory. Mechanical difficulties resulted even thoughthese units were tested in Freon-12, a commercial refrigerant, which reduced the centrifugal stress to one-fourth the stress produced in air at equivalent Mach numbers. The most desirable solution structurally, though impractical in terms of production, appeared to be the construction of the disk, the shroud, and the blades from one piece. The inherently low damping properties of this type of construction, however, further increased the vibration problem.

Vibration investigations were therefore undertaken at the NACA Cleveland laboratory of blade cascades in wind tunnels and of blades operated in the NACA supersonic compressor to determine the effect Of blade thickness, centrifugal force, damping, angle of attack, and surging. The data are presented in the form of curves and photographs. Aerodynamic data from the tunnel tests and compressor performance data are not included herein but are available in references 2 and 3.

VIBRATION INVESTIGATION OF BLADE CASCADES

Inasmuch as the problem of vibration seemed probable in the proposed design, an investigation of blade cascades in 8 supersonic wind tunnel was considered as 8 means of predicting the vibration characteristics of the blades in the compressor. Therefore, while the NACA 24-inch supersonic compressor was being fabricated at the laboratory, such en investigation was made.

Apparatus

Two supersonic blade cascades of one-piece construction were used in the investigation. One, a three-blade cascade was made primarily for developing the machining technique to be used on the final rotor, and the second, a five-blade cascade, was made to investigate vibration end aerodynamic characteristics of the design blade. The blades of the three-blade cascade (four passages) had radial elements with twist along the span of the blades. The five-blade cascade had four passages with the outer blades instrumented with static-pressure tubes to obtain aerodynamic data. The profile of the blade was similar to that of the hub section of the blade to be used in the actual compressor (fig. I). The blades were machined with parallel elements and without twist to produce rectangular air passages. The origin81 design blades for both cascades (fig. 1) had a uniform thickness of 0.024 inch with a wedge section having 8 height of 0.024 inch, the maximum thickness totaling 0.048 Inch.

Two supersonic wind tunnels were used in the investigation, one because it was readily available and the other, which was used later, was designed specifically for the five-blade cascade. The first tunnel was of a closed-throat type in which the three-blade cascade was mounted. The second tunnel was an open-jet type (fig. 2) with the tunnel discharge area coinciding with the passage area between the outer two blades of the five-blade cascade. Two nozzles Were provided for the open-jet tunnel; one nozzle was designed to produce a Mach number of 1.35, the design Mach number of the blade-root section, and the other nozzle was designed to produce a Mach number of 1.62, the design Mach number of the blade-tip section. This tunnel was designed and built in the Applied Compressor Research Section of the Cleveland laboratory.

A special loading, frame was fabricated, which supported the five-blade cascade in the open-jet tunnel snd also tensioned the blades by means of 16 bolts to simulate the centrifugal force imposed on the blades during compressor operation.

A 12-channel amplifier and recording oscillograph were used to obtain permanent records of the signals from the strain gages mounted on the blades. The amplitude-stress relation of the signals was checkedby using two calibration means: (1) a signal of known voltage, end (2) a signal from a similar gage mounted on a cantilever bar that was oscillated through known deflection. The stress at the gage location on the cantilever was determined by direct calculation and by interpolation from strain measurements made with special equipment for this purpose.

Procedure

The first vibration tests were made with the three-blade cascade in a closed-throat tunnel. These tests indicated that the blades were very susceptible to vibration through 8 considerable range of air velocities. The tunnel was designed far a Mach number of 1.8, but the choking caused by the presence of the cascade limited the operation to 8 Mach number slightly above unity. Moreover, the mass of metal supporting the tip and base of the blades could set up bow-and-shock waves, which may have influenced the vibrations. In view of these limitations, the data were questionable, but other preliminary experiments to determine such factors as location of the best strain-gage position and sand patterns Of vibration modes proved valuable in directing further experimentation.

The five-blade cascade was instrumented with strain gages on the three center blades and measurements were taken while the cascade was subjected to the fullest air-velocity range of which the open-jet tunnel was capable. Simultaneous Signals were recorded from the three instrumented blades and analyzed to determine vibratory stress and possible phase relation between vibrations of adjacent blades. All vibration signals were corrected for the frequency response of the recording equipment.

The first part of the investigation of the blades was made with the thickness of the uniform section increased to three times the original design thickness, which, including the wedge, gave a maximum thickness of 0.096 inch. Additional runs were made after the uniform section had been progressively machined to give maximum thicknesses of 0.072 inch, 0.060 inch, and 0.048 inch. Before each run, the blades were uniformly tensioned by means of the 16 bolts in the loading frame. The tension stress was measured by expendable gages located on the mid span at the leading and trailing edges and at the mid-chord position of the three center blades to give the desired uniform blade stress. The 0.096-inch-thick blades were stressed to 45,000 pounds per square inch, the limit imposed by the bolt strength. The blades of the other thicknesses were loaded to 60,000 pounds per square inch, the average centrifugal blade stress at rated compressor speed.

Vibration surveys were conducted on the blades in both the tensioned and untensioned conditions using both nozzles in each condition. Highest vibratory stresses were observed at a Mach number of approximately 0.8; consequently, tests on the 0.048-inch-thick blades were conducted once more with a tension of 23,000 pounds per square inch, corresponding to the centrifugal stress at the speed that would produce air speeds of a Mach number of 0.8. All Mach numbers reported are of the air velocity at the entrance to the blade cascade.

The damping of the one-piece construction employed is inherently very low but was greatly increased by the supporting and loading of the cascade. It was therefore necessary to evaluate this added variable. The increased damping effect was determined by the analysis of vibration die-away curves made of the cascade while suspended by thin wires, while in the loading frame, and while mounted in the loading frame in the tunnel.

In order to verify the necessity of the shroud, the tips of the blades of the five-blade cascade were cut free from the shroud and again mounted within the open-jet tunnel and subjected to 8 range of air velocities.

Discussion and Results

The data presented are results of the five-blade cascade tests and must be regarded as having limited accuracy because of the nature of the blade vibrations and the uncontrollable factors affecting these vibrations. For example, the amplitude of vibration is very irregular as illustrated by a representative signal shown in figure 3. The maximum amplitude on the record is the one used for further analysis and, during the period of exposing the film, usually 1/2 second, peak vibrations may not have been present. In order to obtain 8 greater degree Of accuracy, considerable film would have to be exposed for each operating condition. Furthermore, the location of the gage proved to be extremely critical and it was impractical to reuse the same gage for all runs. The data seemed to consist of scattered groups of points; however, by averaging several points for each condition, very definite trends Were indicated.

The galvanometer elements of the recording oscillograph respond differently to different frequencies, and therefore 8 correction factor was applied to the observed stress to eliminate the effect of the difference between blade frequency and calibrator frequency. A second correction factor involving total damping was applied to the final data. An analysis Of the vibration die-away curve showed that by supporting the blade cascade in the loading frame the damping increased and thereby reduced the vibration amplitude. The proposed one-piece compressor construction, it is believed, has damping properties closely resembling those Of the one-piece blade cascade suspended independently of the loading frame.

The aerodynamic damping inherent in the system could not be evaluated. This type of damping may help to decrease vibration amplitude in the compressor or it may increase amplitude by becoming negative, 8s in the case Of classic flutter. If the aerodynamic damping present in the tunnel tests is nearly equal to that which will occur in the compressor in service, the vibration characteristics determined in the tunnel should give 8 good indication Of those present in the compressor.

The highest vibratory stress measured in the tunnel With the five-blade cascade was a single amplitude stress of ±39,600 pounds per square inch. This peak stress was observed at a total thickness of 0.060 inch with the blade untensioned, With the high-velocity nozzle, and at 8 Mach number of 0.8. With all data points averaged (fig. 4) for both the tensioned and untensioned conditions regardless of Mach number, the stress increased with reduction in blade thickness and therefore the highest measured stress (39,600 lbs/sq in.) should have occurred in the 0.048-inch-thick, not the 0.060-inch-thick blade. The explanation for the apparent discrepancy is present in figure 5 on the records for the 0.048-inch-thick blade. A higher-mode vibration was being excited only in the untensioned 0.048-inch-thick bled8 along with the fundamental mode, which predominated in the other thicknesses end loading conditions. The higher mode absorbed pert of the energy exciting the fundamental mode and thus reduced the amplitude of the blade vibration.

The effect of the two nozzles on the vibratory stress is shown in figure 4. The nozzle designed for a Mach number of 1.62 produced amplitudes end stresses approximately twice as large 8s those produced by the nozzle with a Mach number of 1.35. Flow conditions in the actual compressor may be much better than those in the tunnel, because the inlet section is designed for the air velocities expected and flow will be subsonic until it reaches the blades. Because nozzle characteristics have such 8 decided effect on blade vibrations, it can be doduced that the vibrations in the compressor blades might be lower than in the cascades.

Blade tension has the greatest effect on the blade vibration, as is shown by figure 6. It can be seen that in the untensioned blade of 0.096-inch thickness, practically no stress amplitude was observed; this fact is probably explained by the difference in damping due to the mass ratio of the blade and the remainder of the system, Also, the blade stiffness was increased because of its thickness, which affects the transfer of vibration energy. When the blade was tensioned, the added damping was removed and a higher stress resulted. The damping correction factors were based only on results of the thinnest blade and consequently did not include correction for mass ratio and rigidity effects.

All runs, regardless of blade thickness, blade tension, or nozzle, showed similar vibration response to the Mach number range. Representative curves of stress plotted against Mach number are shown in figure 7. The gage on blade 3 was better located with respect to maximum vibratory stress then those on blades 1 and 2 and the difference in stress is not necessarily an indication that this blade vibrated with higher amplitude. In some of the first runs, blade 1 showed higher stress than was simultaneously indicated far the other two blades. The results of the more favorable location of the strain gage (on blade 3) is also shown in figure 8. This figure was plotted from data obtained with a static-strain measuring bridge, the Strain present having been produced by mechanically deflecting the blade in the center of the span at the leading edge. From this curve, the vibration amplitude present can be estimated for any observed stress in the 0.04%inch-thick blade.

The mode of vibration being excited is a slightly distorted first torsional mode, as is indicated by the sand pattern (fig. 9). The numerical frequency of the illustrated mode of vibration is less than the frequency of the first bending mode. The blade is therefore particularly susceptible to flutter because the lowest natural frequency is generally the easiest to excite, and flutter generally involves torsional vibrations. The leading and trailing edges in the center of the span possess the highest amplitude and the highest stresses are located near the blade root and tip at the leading and trailing edges. This fact is substantiated by the results of the experiments on gage location. Unfortunately, the strain gage has finite size, though small, and therefore cannot be located at the point of maximum stress, which is probably *at* the edge of the blade. Furthermore, the gage averages the stress over the small area.

The possibility that contraction and expansion of the passage between adjacent blades might be the cause of the apparent vibration Was investigated. The possibility was eliminated because simultaneous signals from adjacent blades were not 180° out of phase as would be required to produce this result. On the contrary, there appeared to be no relation between the phases or the amplitudes of vibration of adjacent blades.

The effects of blade tension and blade thickness on frequency are shown in figures 10 and 11, respectively. The increase of the first torsional mode frequency caused by applying tension is somewhat greater than Was expected.

The significance of all the results obtained in the tunnel runs is best illustrated in figure 12. The values for the maximum allowable single-amplitude vibratory stress were calculated from the modified Goodman diagram corresponding to the rotor material hardness of a Brinell hardness number of 363 in reference 4. The abscissa, rather than being marked off in mean working stresses, which in the case of the compressor are centrifugal and bending stresses, is graduated in Mach numbers, thus enabling immediate correlation with tunnel data. None of the observed peak stresses exceeded the maximum allowable limits. Therefore compressor blades of the proposed design should be able to attain full operating speeds without blade failure caused by vibratory stresses, provided that no stress-concentration factors are introduced during fabrication or operation. All peak stresses observed for

0.072-inch and 0.096-inch thickness were less than 20,000 pounds per square inch and consequently do not appear on figure 12.

During the tunnel investigation of the unshrouded blades, the strain gages quickly failed. Visual inspection of the blades showed the leading edges to be vibrating with tip amplitudes of approximately 1/2 inch. Observation of the motion under stroboscopic light showed the blade to be fluttering violently in the first torsional cantilever mode ofvibration. When the model was removed from the tunnel after a short period of such operation, one of the blades was cracked over 30 percent of the chord length from both the leading and the trailing edges (fig. 13). In figure 13(a), the progression of the crack through the strain gage indicates proper location of the gage to measure maximum vibratory stresses.

VIBRATION SURVEY OF COMPRESSOR

In view of the results from the tunnel runs, the compressor rotor was completed as originally proposed except that the blades were made 0.060 inch Instead of 0.048 inch thick. The aerodynamic tests conducted in the tunnels simultaneously with vibration surveys indicated satisfactory operation with the thicker blades. Also, vibration conditions were improved and blade stresses caused by the centrifugal shroud load were reduced by use of the thicker blade. Extreme caution was exercised in finishing the rotor to remove all nicks end scratches that might cause failure by introducing stress concentration.

Apparatus and Procedure

The compressor (reference 3) was instrumented with strain gages to warn against sustained operation at 8 condition of excessive stress not duplicated in the free-jet tunnel. Two active gages, attached at the leading edge and near the base of diametrically opposite blades, were used to measure vibratory stresses. Balance gages forming the inactive bridge arms of the electric measuring circuit were mounted under the overhanging rim of the disk, as shown in figure 14. An 18-ring, slip-ring assembly was used to transfer the strain signal from the rotor to the recording equipment. Two rings per conductor were utilized to minimize elip-ring interference.

The rotor was investigated in a 9000-horsepower variable-component compressor setup. The inlet and the discharge of the compressor were connected to laboratory facilities that permitted thorough investigation of all operating variables such as air temperature, mass flow, air density, and inlet and discharge pressures.

The vibration survey was conducted by continually observing the vibration traces on oscilloscopes and by recording the stresses indicated at test points by means of a recording oscillograph. At each test point, the records were analyzed to determine the stress amplitude, frequency of vibration, and the rotor speed of the compressor. The tensile stress in the blade, due to centrifugal force, was measured by utilizing an active strain gage and a bridge circuit sensitive only to the steady-state stress in order to verify the calculated high stress and possible stress concentration at the blade root.

The Mach numbers reported are of the air flow relative to the blades at the rotor entrance.

Discussion and Results

The results of wind-tunnel investigation on cascades were verified by those obtained with the compressor rotor. The compressor investigation showed that the blades are subjected to flutter vibrations throughout the complete speed range of the unit. No critical speeds were noted, mainly because the compressor was installed without inlet guide vanes or diffuser vanes. When these vanes are incorporated into the unit, they may introduce pressure variations and end turbulence causing strong vibrational excitation. The data indicate that the vibration amplitude was influenced principally by increasing the angle of attack in either positive or negative directions (fig. 15). The operating variables previously mentioned had no decided effect on the vibration, although a sustained vibration was noted at a Mach number of about 0.7, which may be close to the critical Mach number for the airfoil design. The formation of shock waves within the blade passage produced no noticeable effect on the vibration amplitude. When a condition of surge was produced in the compressor, the blades fluttered violently in a pulsating manner. A typical record of this vibration is shown in figure 16. The blades vibrated in the fundamental torsional mode, the frequency of which increased with rotor speed, as shown in figure 17. Because the blade frequencies are nearly equal throughout the rotor, a beat frequency can occur and a typical example of this beat frequency is shown in figure 18. Figure 18(a) shows a record of the blade vibration-e the compressor was in operation. Figure 18(b) shows a vibration as a result of air passing through the stationary rotor. At this condition severe vibrations have been observed and, in view of the data, can be attributed to the blades being situated at high angles of attack.

The stress amplitude was calculated from the maximum signal height on the record. The maximum single amplitude stress recorded was ±23,250 pounds per square inch at 8 rotor speed of 7770 rpm, 8 Mach number of 0.714, and an effective angle of attack of 13° 36'. However, the vibratory stresses encountered at higher speeds are of more significance because of the higher superimposed tensile stresses. A single amplitude stress of ±14,500 pounds per square inch was recorded at 15,600 rpm at surge, although at the same speed below eurge, stresses of 3000 to 4000 pounds per square inch were encountered. A plot of the measured tensile stress due to centrifugal force is presented in figure 19 together with the calculated stress. The difference between the measured and the calculated stress is due primarily to the stress concentration at the gage location and slight strain of the inactive gages as well 8s a temperature gradient between the rotor rim and the blade root. Nevertheless, it can be seen that the stress is high and that the superimposed vibratory etrese is limited.

SUMMARY OF RESULTS

Fran a vibration investigation of a supersonic blade cascade in a supersonic open-jetwindtmnel and of blades operated in a compressor, the following results were obtained:

1. Cascade test data showed good correlation with those Of blades in the compressor.

2. Peak vibratory stresses of ±39,600 pounds per square inch, single amplitude, at a Mach number of 0.81 and ±23,250 pounds per square inch, single amplitude, at a Mach number of 0.714 were observed in the wind tunnel and the compressor, respectively.

3. Increasing blade thickness or tensioning reduced vibratory stress.

4. At a Mach number of 1.41, the maximum recorded stress was ±14,500 pounds per square Inch, single amplitude.

5. High effective angles of attack, negative or positive, induced vibrations at all compressor speeds.

6. Severe vibration of the blades occurred when air was passed through the stationary compressor.

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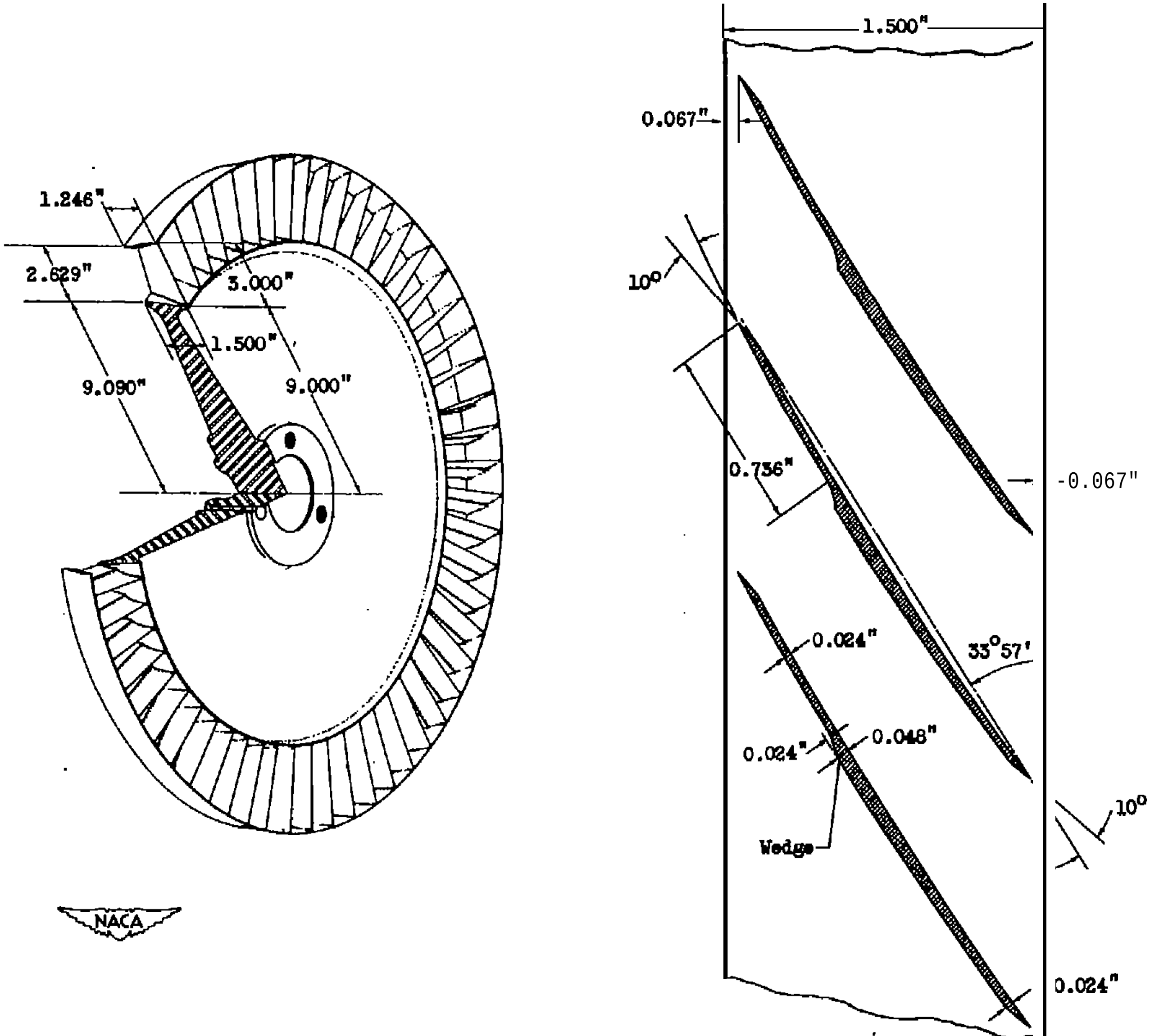


Figure 1. - Axial-flow supersonic compressor showing blade arrangement and blade profile.

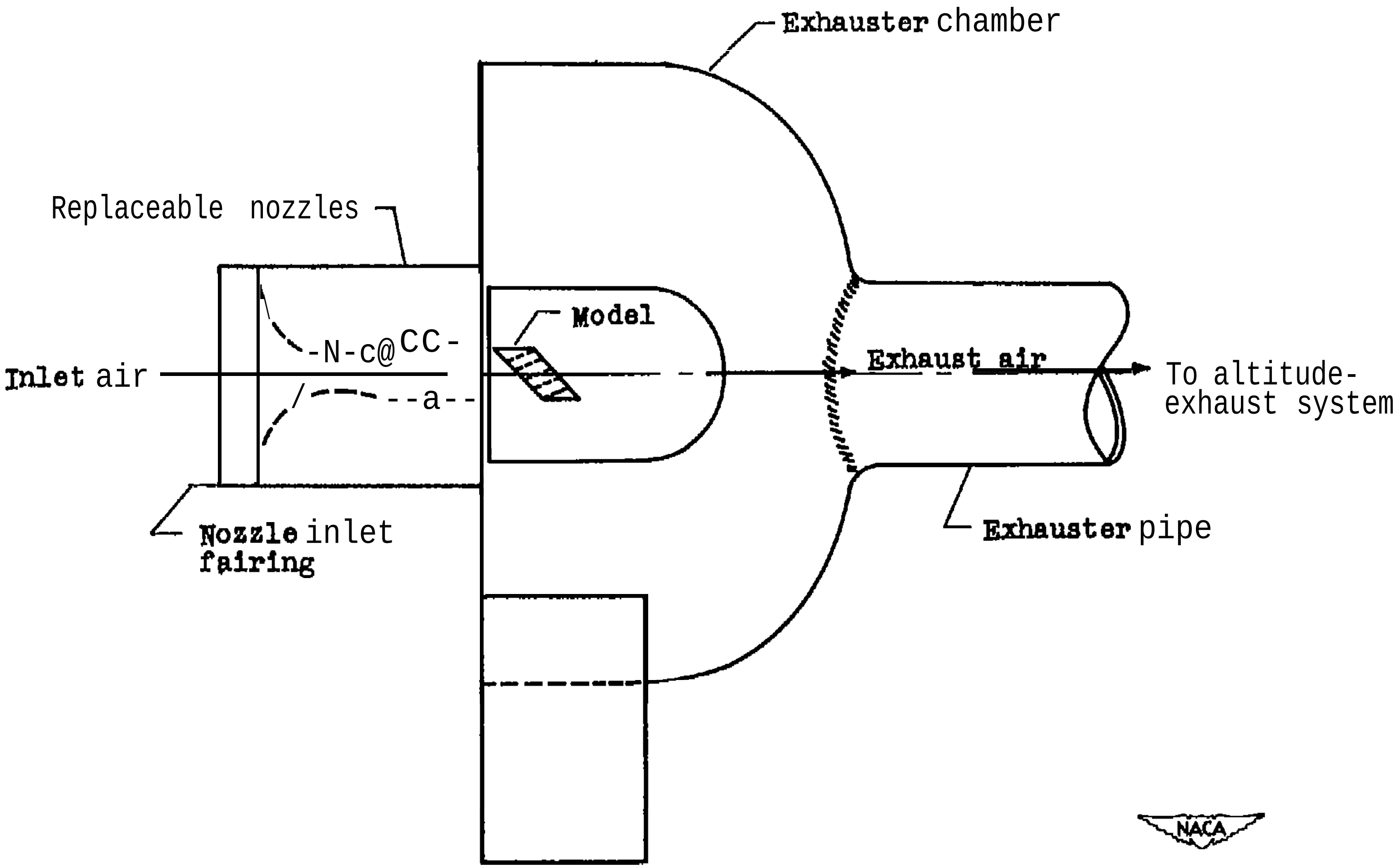


Figure 2. - Supersonic open-jet cascade tunnel.

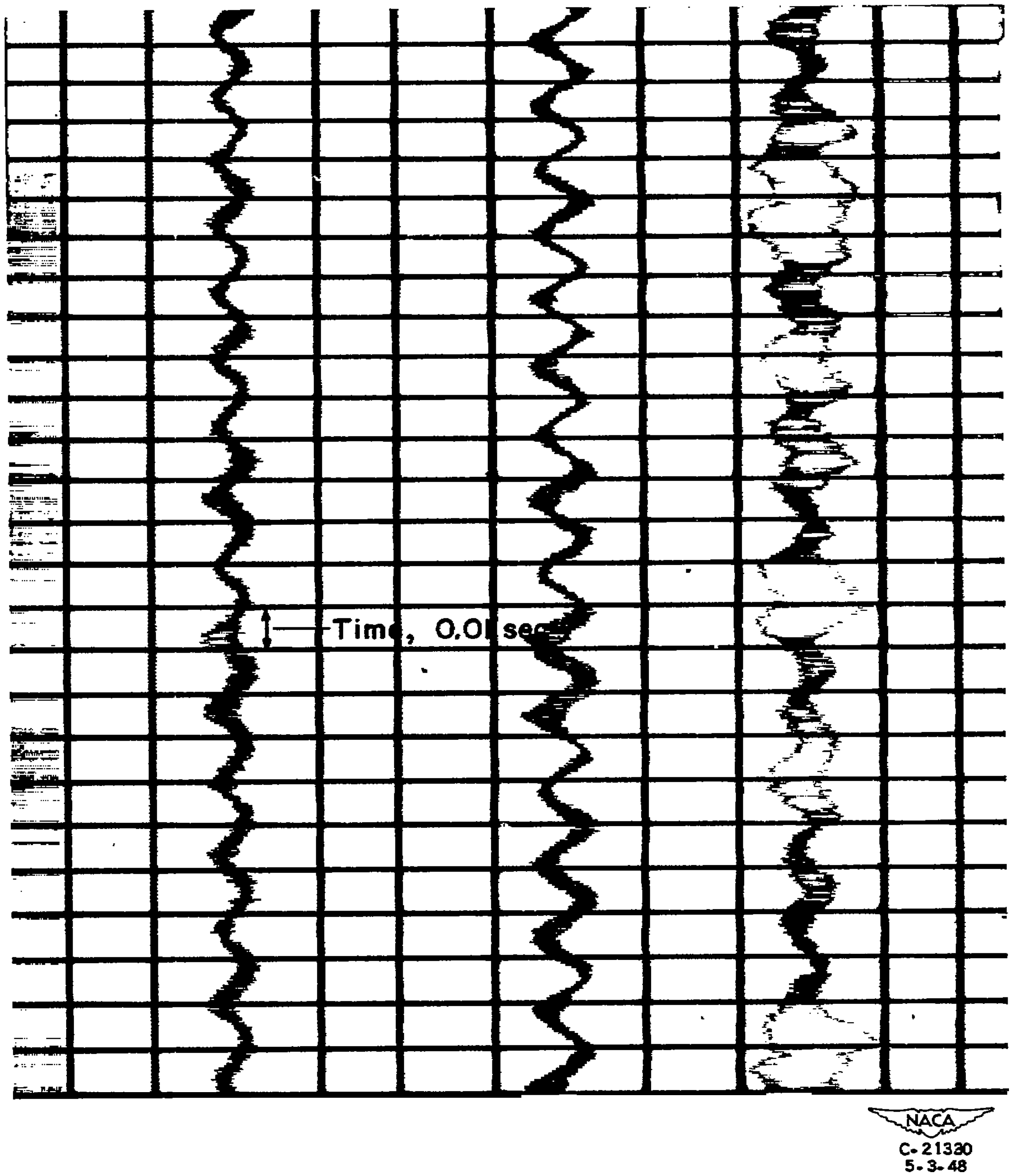


Figure 3. - Oscillograph record showing typical strain-gage signal of cascade-blade vibration.

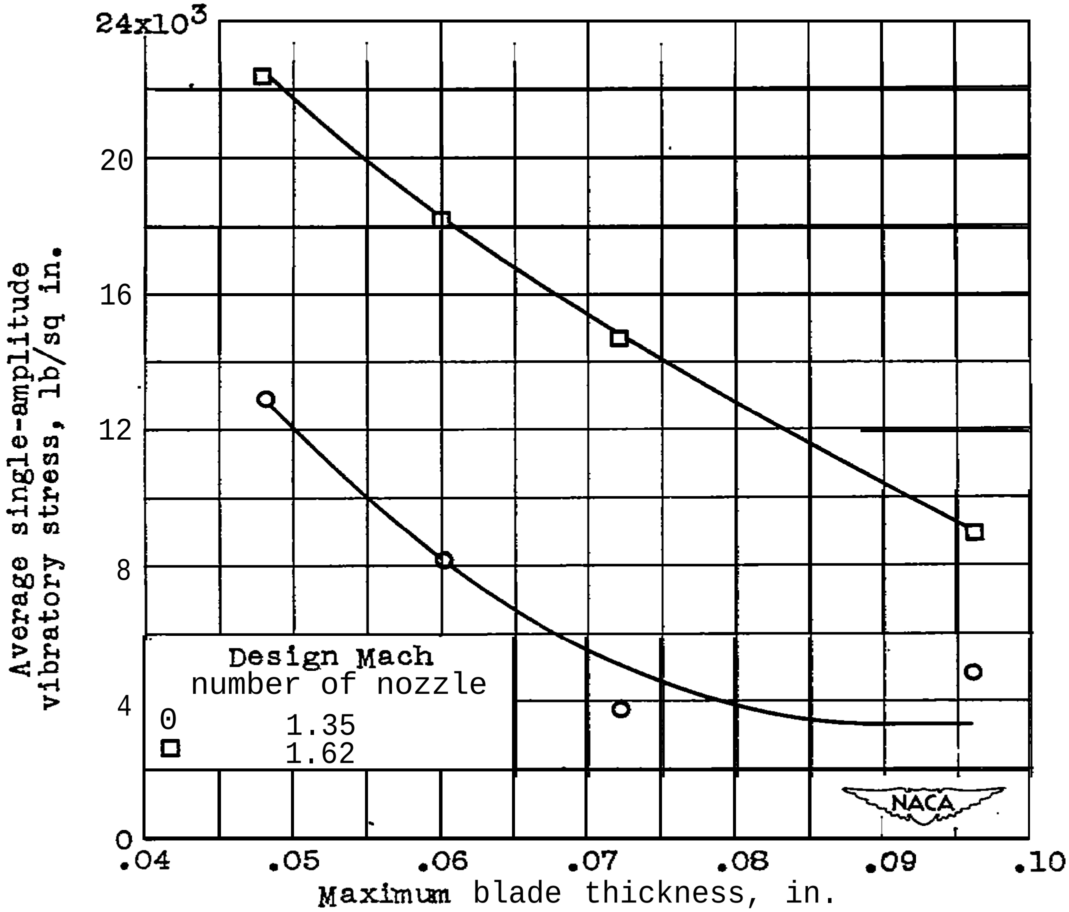


Figure 4. - Effect of blade thickness on average vibratory stress of tensioned and untensioned cascade blades for nozzles with design Mach numbers of 1.35 and 1.62.

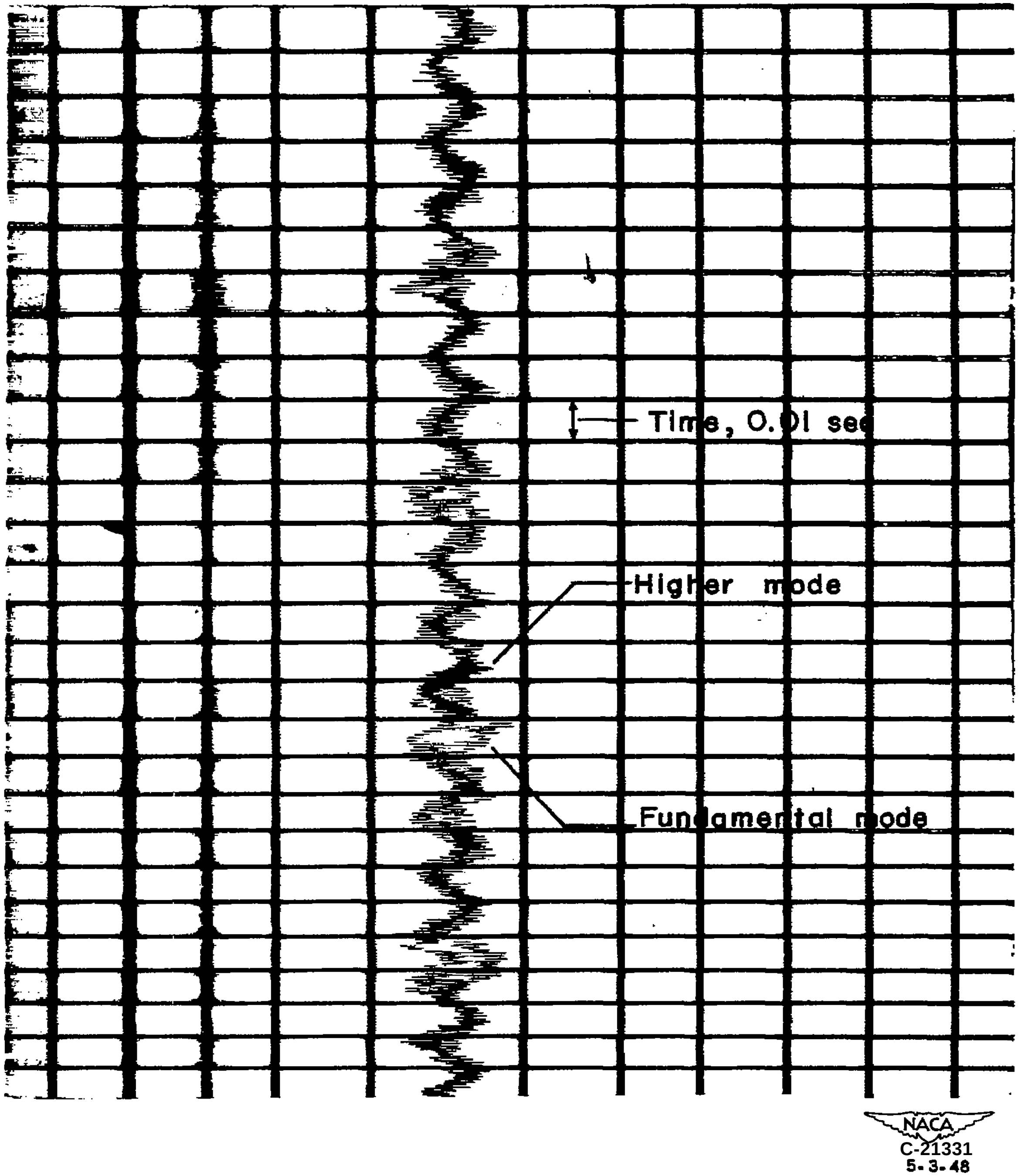


Figure 5. - Oscillograph record for cascade blade showing higher-mode vibration superimposed on fundamental-mode vibration. Blade thickness, 0.048 inch.

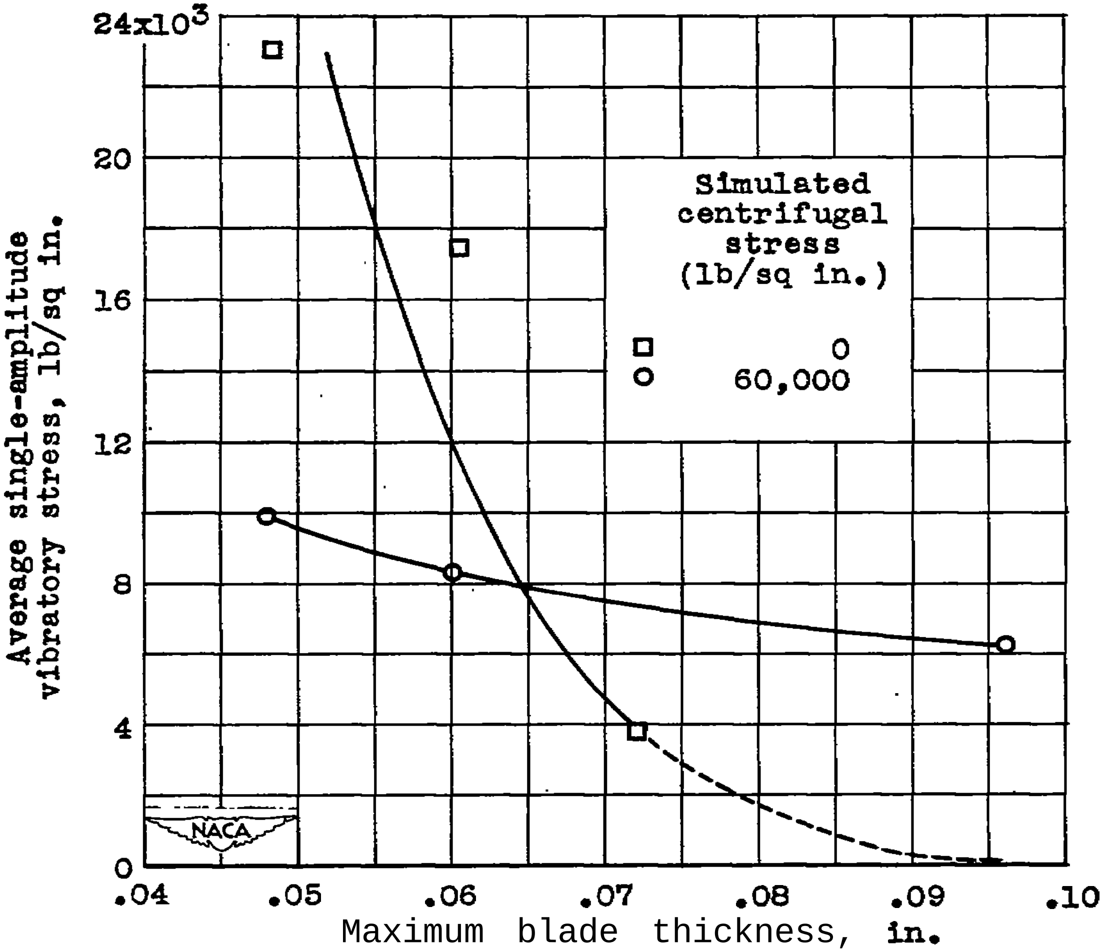


Figure 6. - Effect of blade tension on vibratory stress for various blade thicknesses at various air velocities for cascade blades,

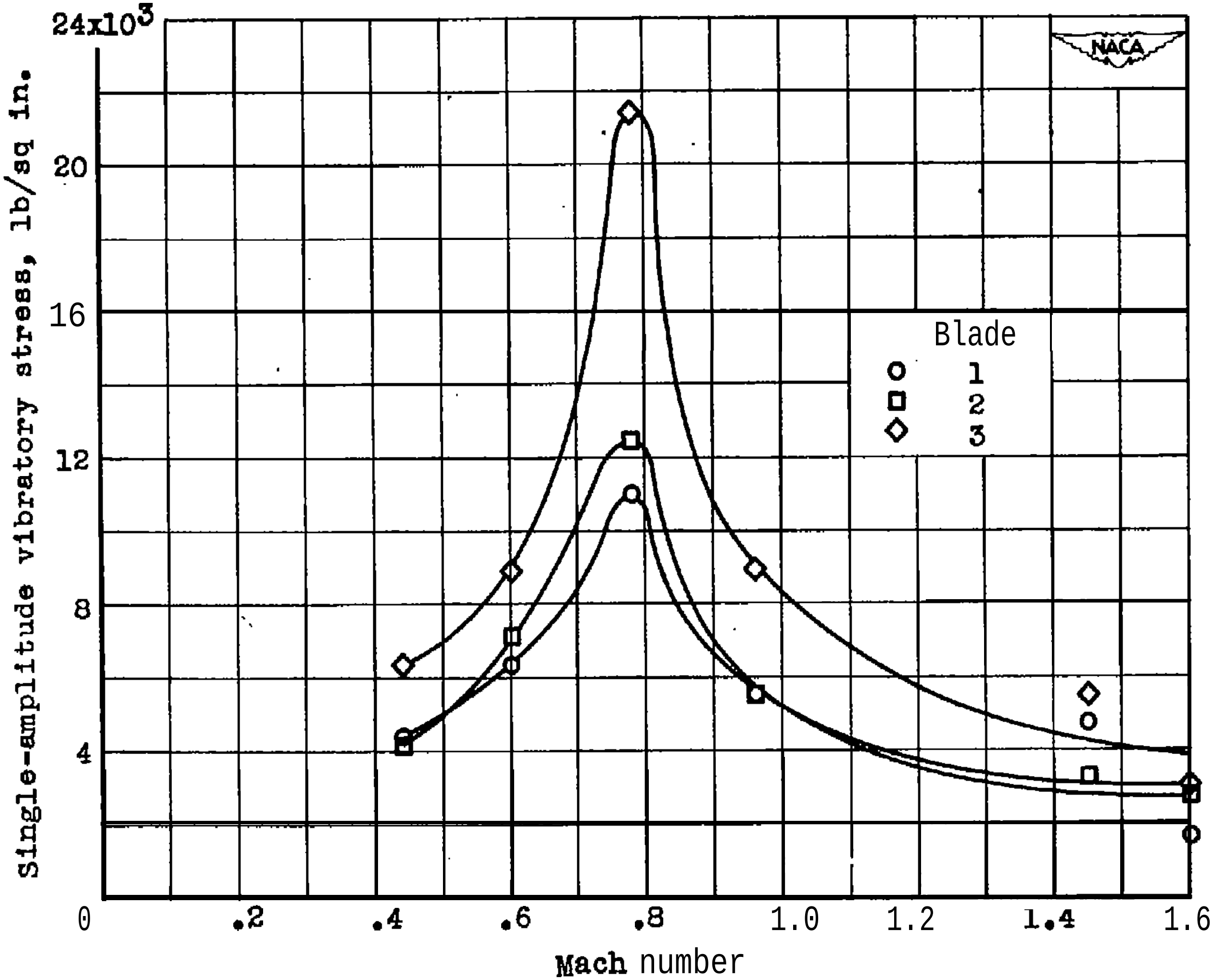


Figure 7. - Effect of Mach number on vibratory stress showing peak stress at Mach number of approximately 0.8 for cascade blades. Blade thickness, 0.048 inch.

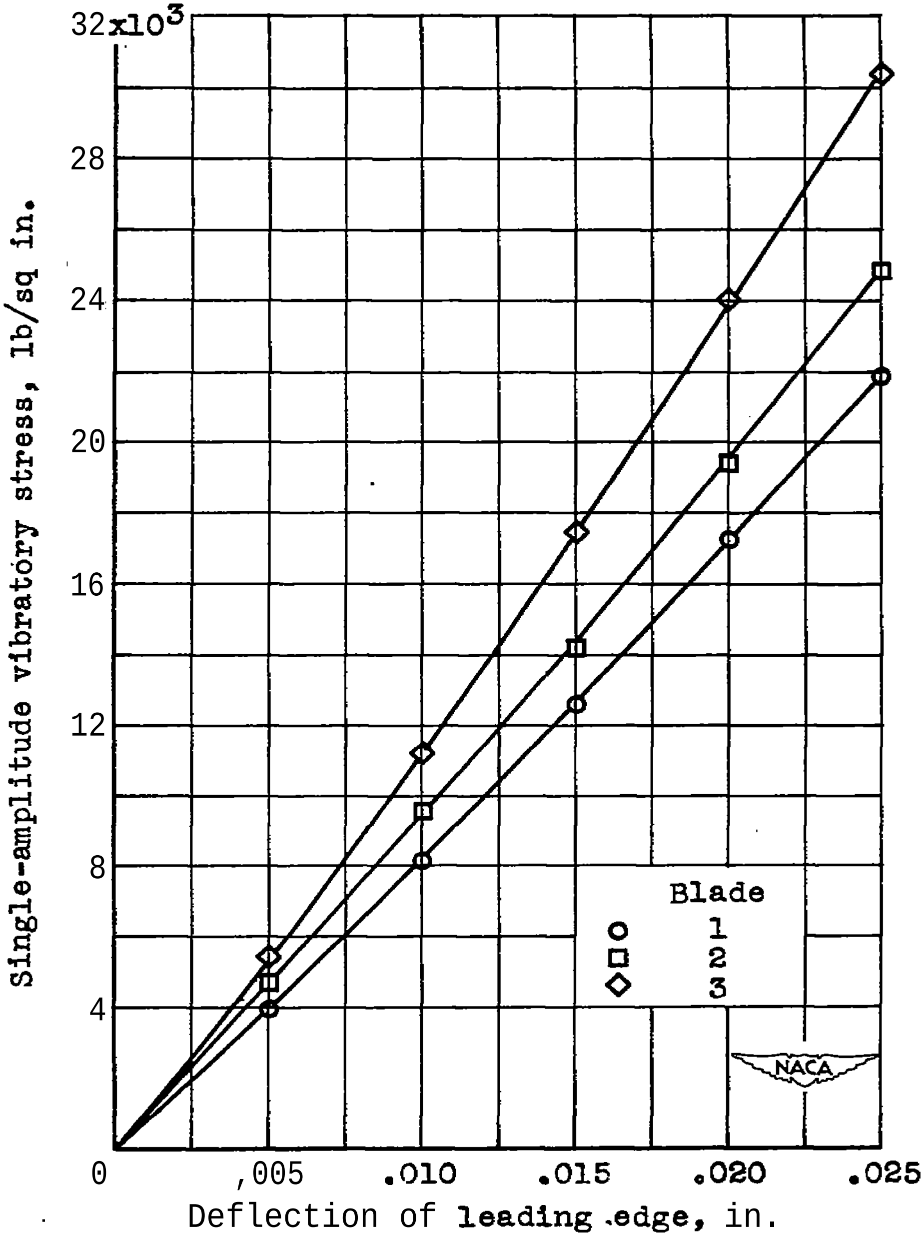


Figure 8. - Stress due to static deflection of leading edge of supersonic-compressor blade in cascade. Blade thickness, 0.048 inch.

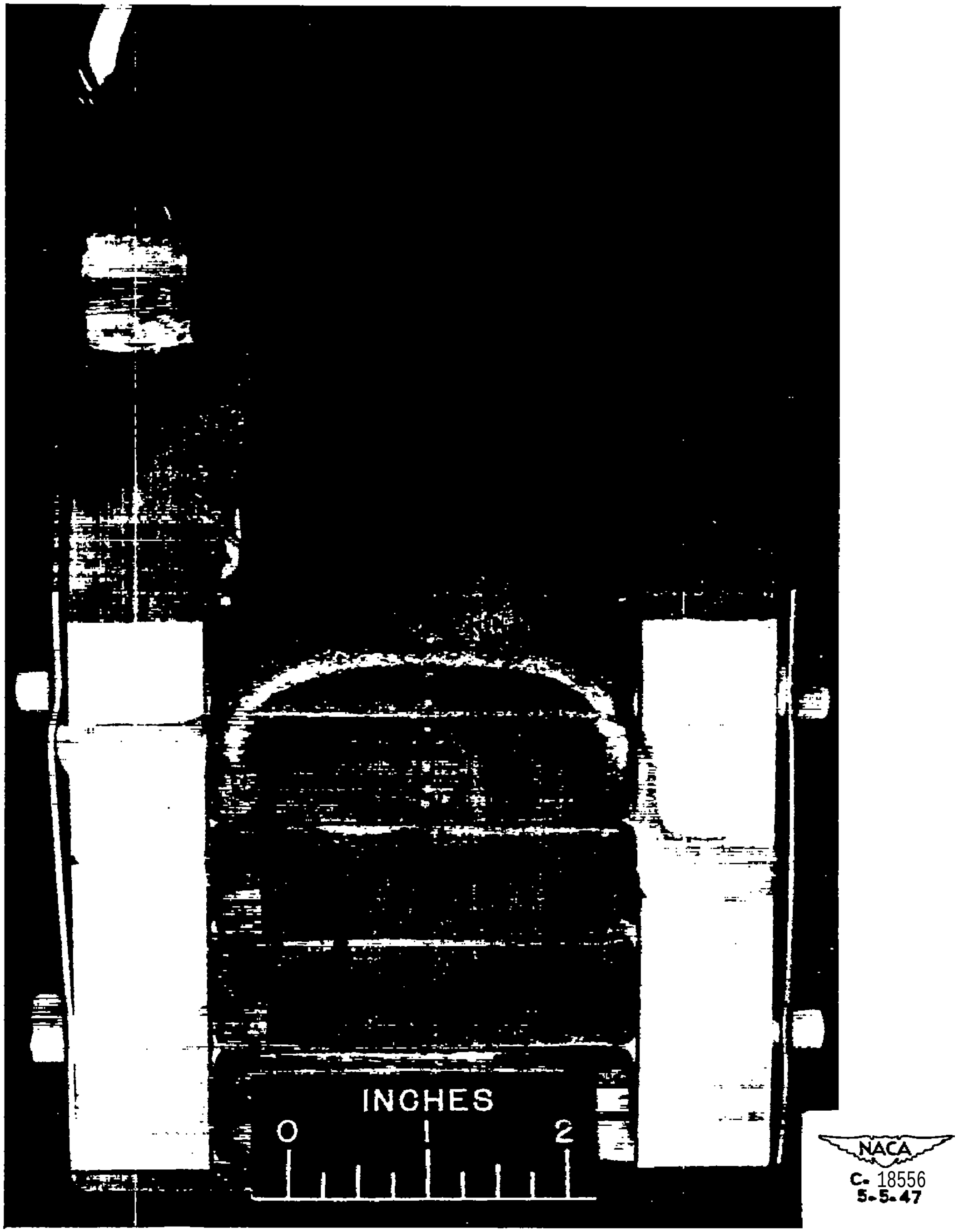


Figure 9. - Sand pattern showing distorted first torsional mode of vibration in supersonic-compressor blade in cascade.

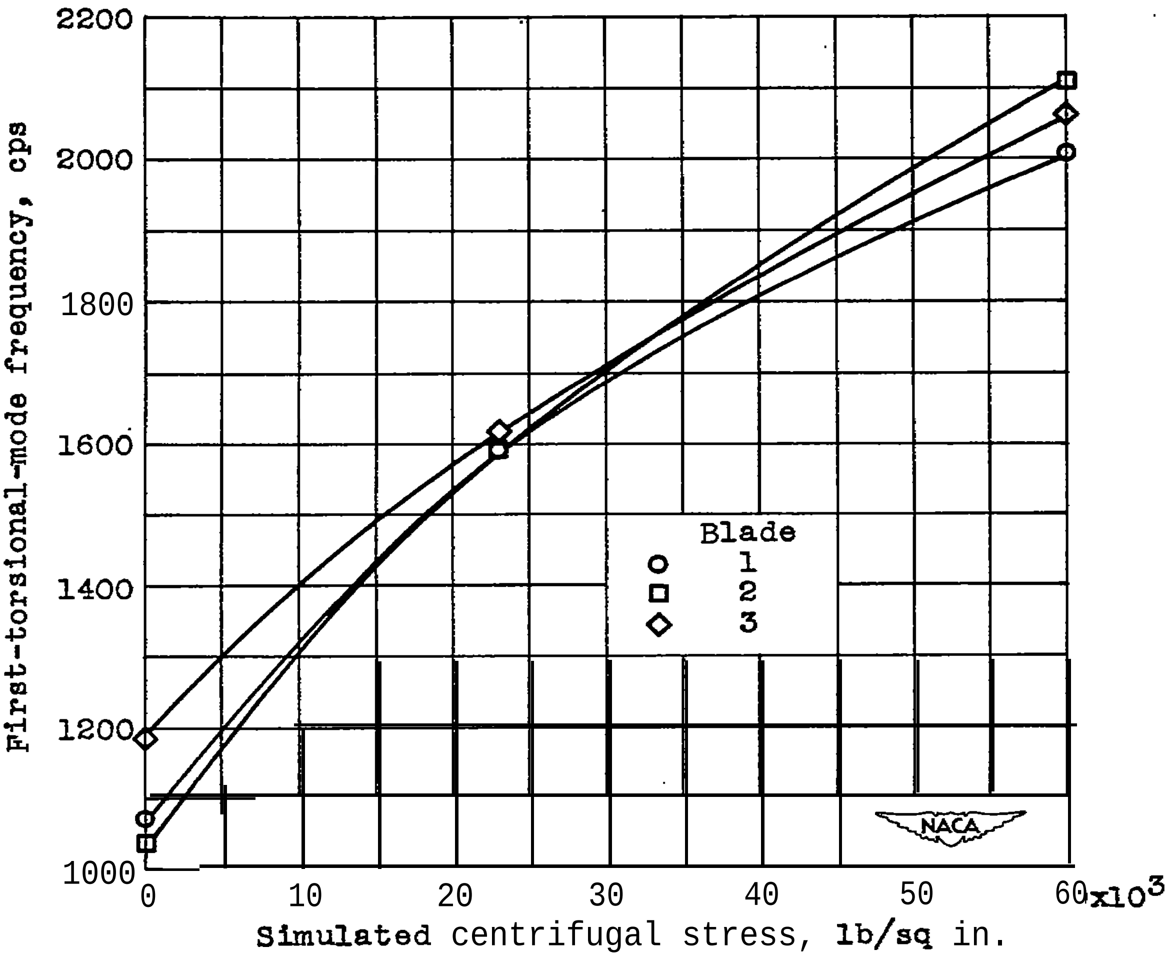


Figure 10. - Effect of simulated centrifugal stress on cascade-blade frequency. Blade thickness, 0.048 inch.

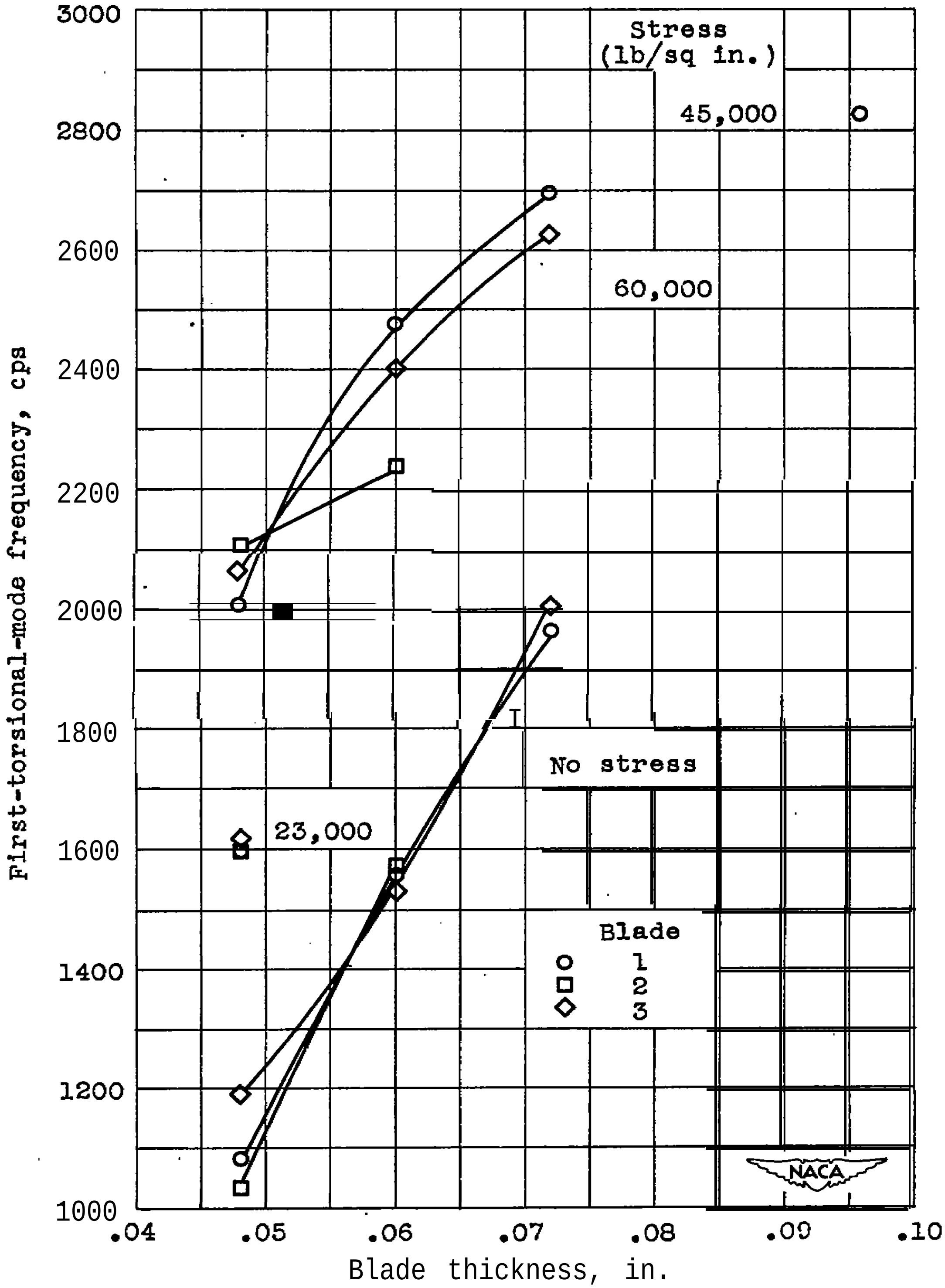


Figure 11. - Effect of blade thickness on blade frequency for -cascade blades.

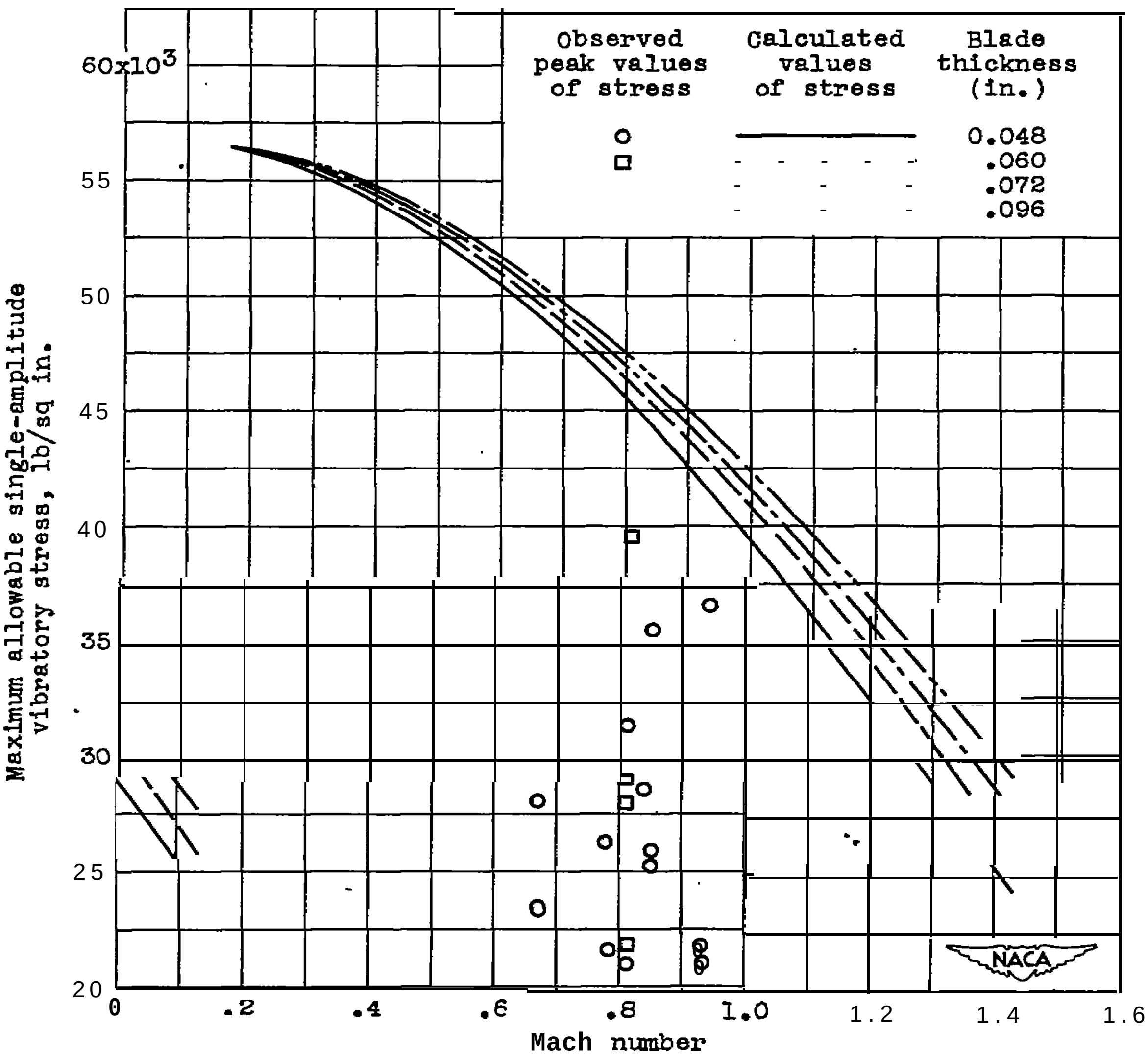
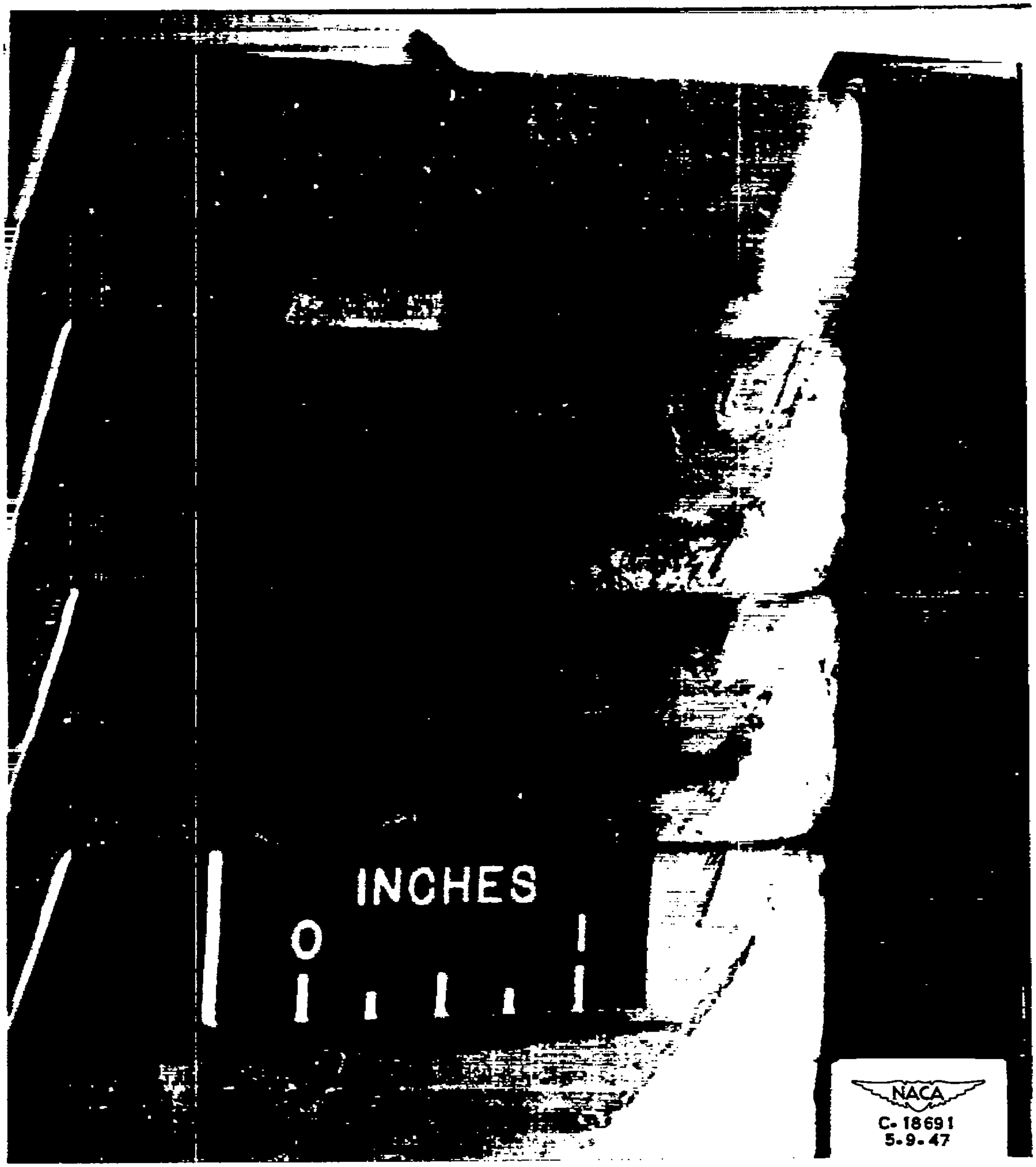
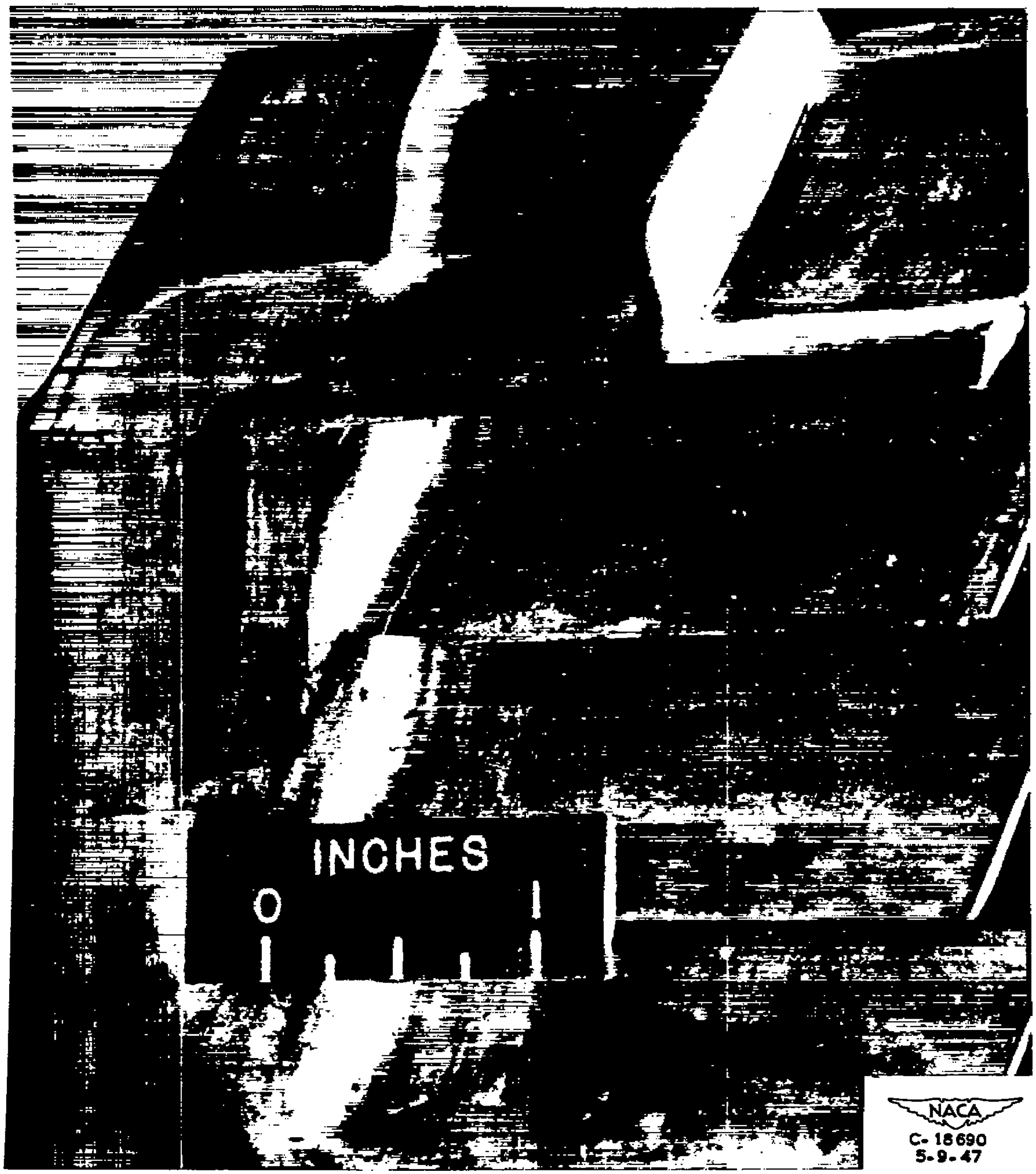


Figure 12. - Allowable vibratory blade stress at given Mach number for cascade blades.



(a) Leading edge.

Figure 13. - Failure of unshrouded blade in cascade.



(b) Trailing edge.

Figure 13. - Concluded. Failure of unshrouded blade in cascade.

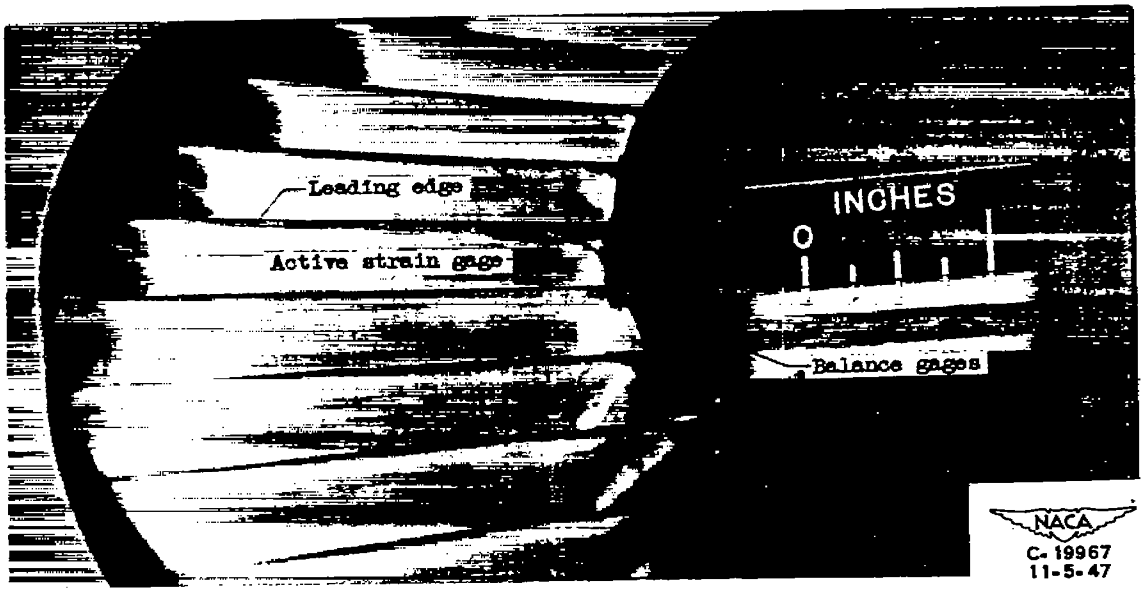


Figure 14. - Strain gages mounted on supersonic compressor.

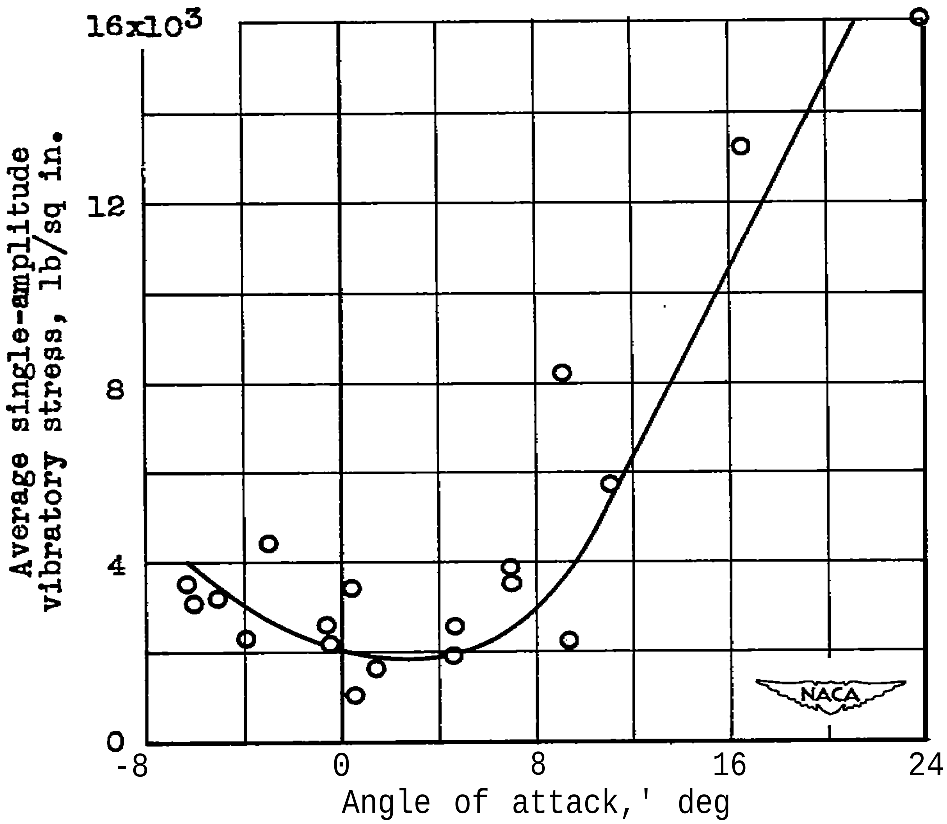


Figure 15. - Effect of angle of attack of super-sonic-compressor blade on vibratory stress.

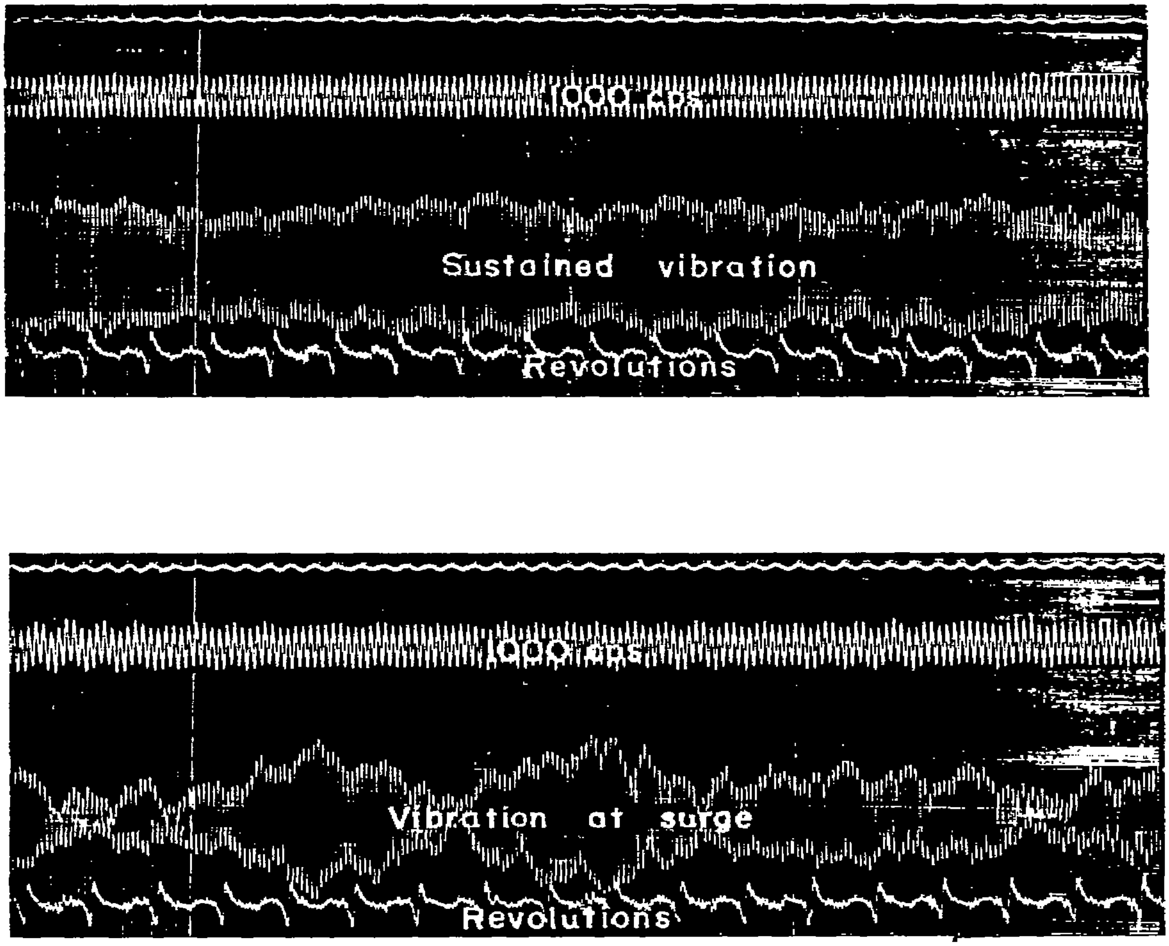


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Figure 16. - Oscillograph records showing typical strain-gage signals of supersonic-compressor blade.

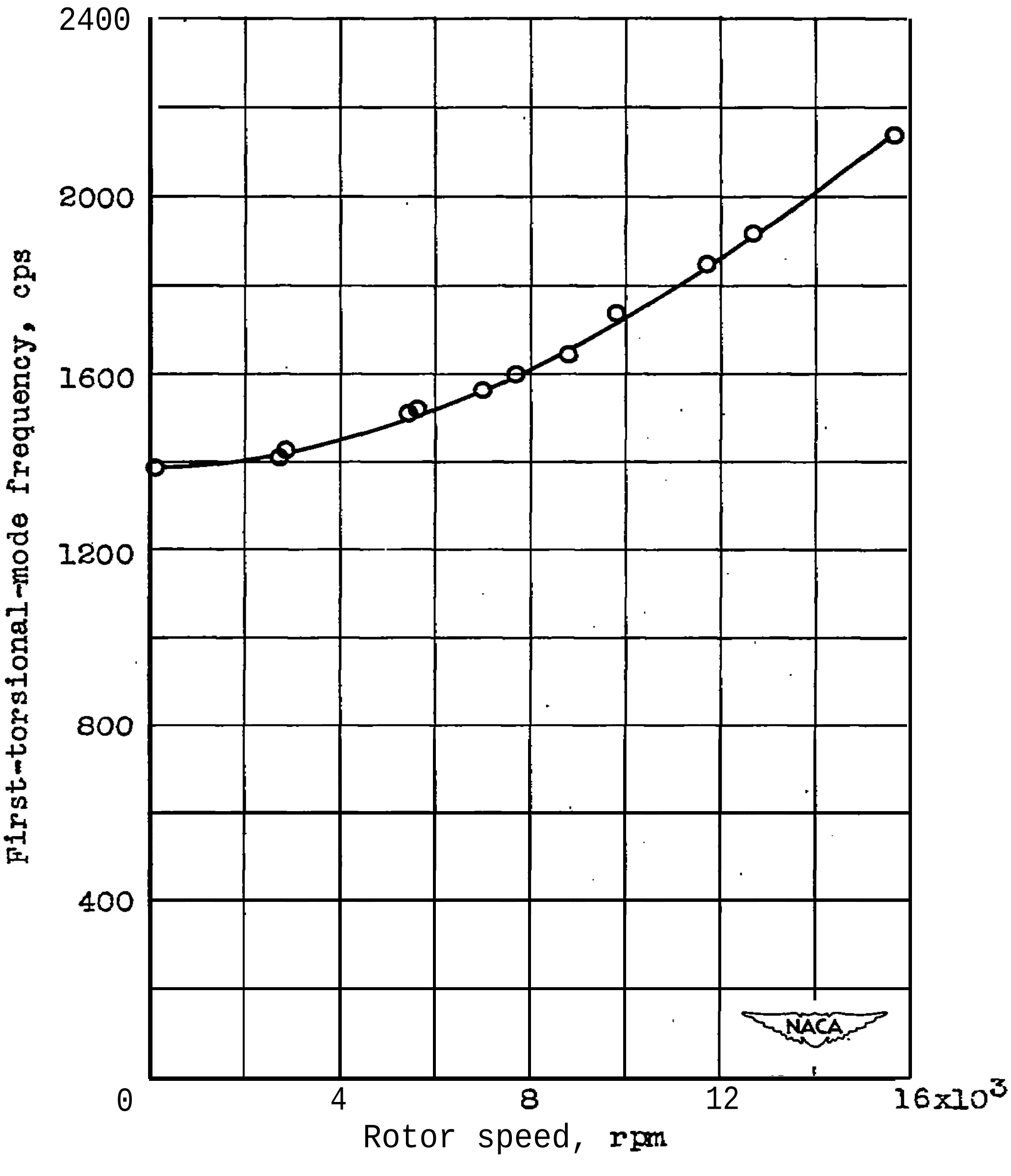
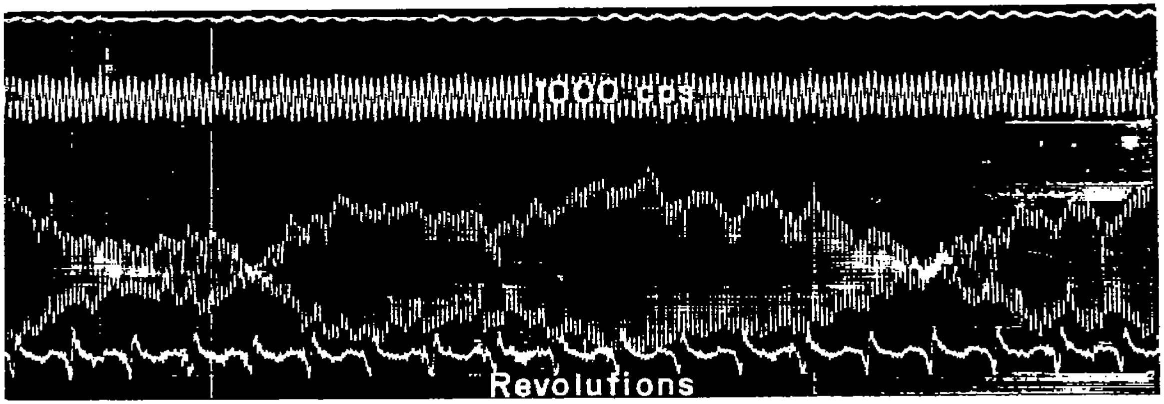
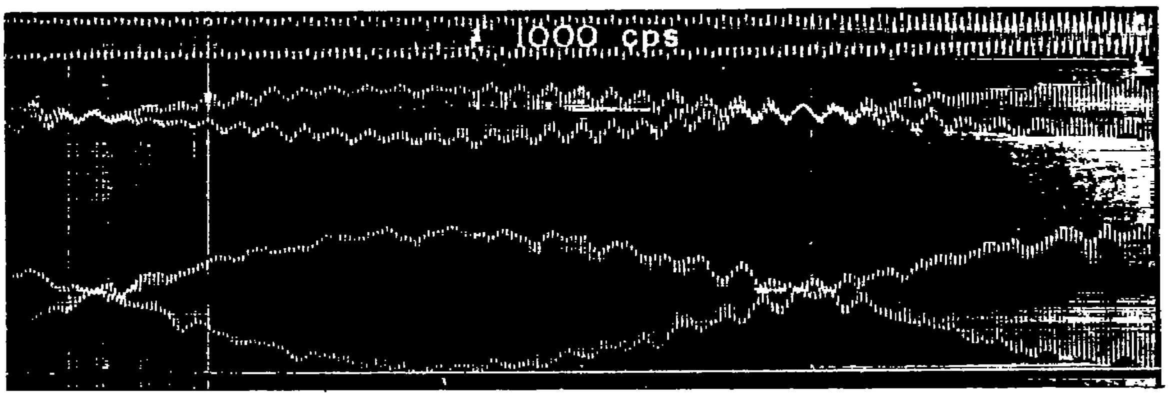


Figure 17. - Effect of centrifugal force on supersonic-compressor-blade-frequency.



(a) Strain-gage signal of flutter at 6950 rpm.



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(b) Strain-gage signal of flutter due to air passing through stationary rotor blades.

Figure 18. - Oscillograph records of blade vibration showing possible effect of beat frequency.

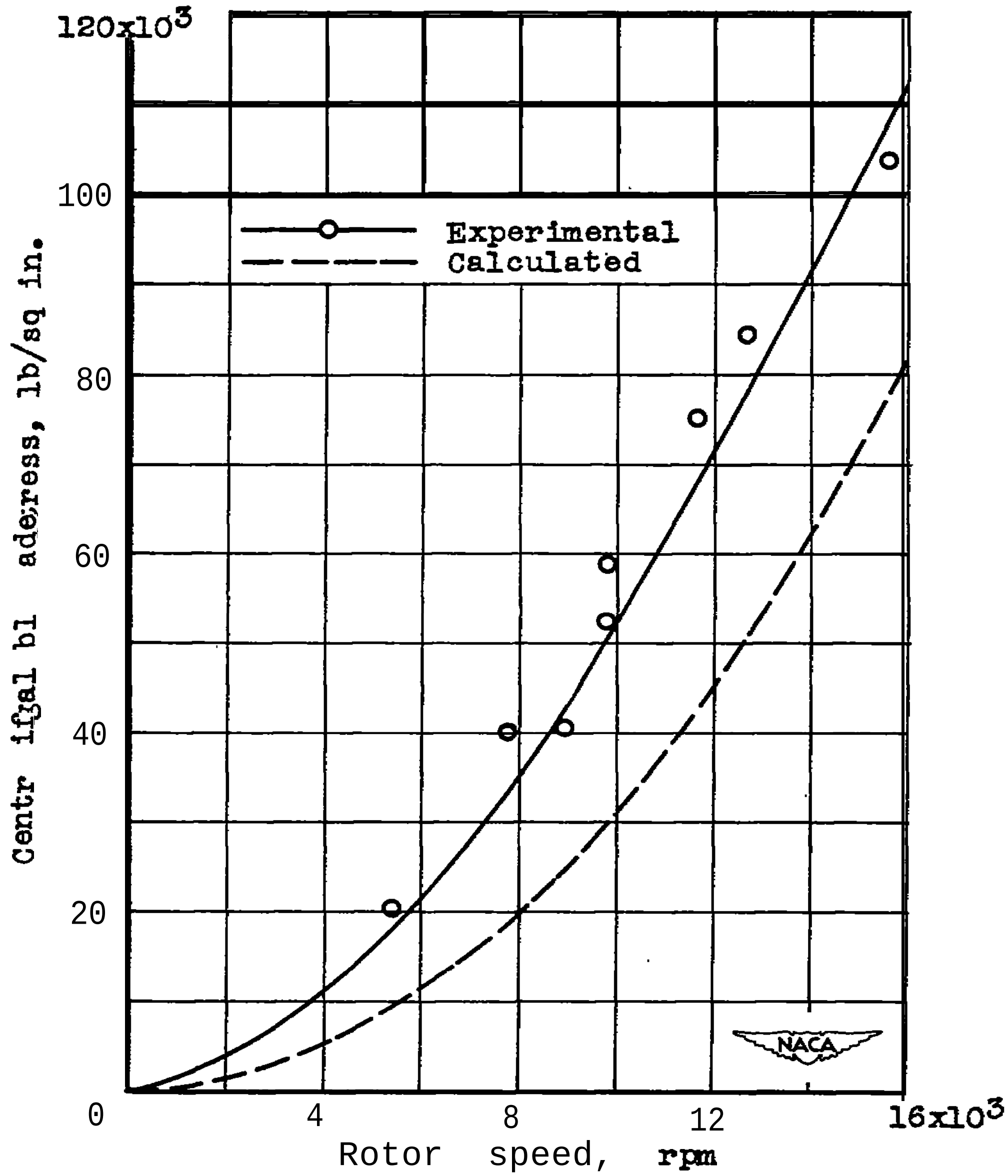

Figure 19. - Effect of centrifugal force on supersonic-compressor-blade stress.